MULTIMEDIA UNIVERSITY

STUDENT ID NO

MULTIMEDIA UNIVERSITY

FINAL EXAMINATION

TRIMESTER 1, 2018/2019

EME4016 – HEAT TRANSFER

(ME)

18 OCTOBER 2018
9.00 a.m. - 11.00 a.m.
(2 Hours)

INSTRUCTIONS TO STUDENTS

1. This Question Paper consists of nine pages including the cover page and Appendix.

2. Answer ALL questions. Each question carries 25 Marks and the distribution of the Marks for each question is given in brackets [ ].

3. Write all your answers in the Answer Booklet provided.

Question 1

A 4-cm thick large steel plate ($\rho = 7832$ kg/m^3, $c_p = 434$ J/kg ·°C and $k = 63.9$ W/m ·°C) which is initially at a uniform temperature of 25 °C passes through an oven at 800 °C. The surface temperature of the plate as it comes out of the oven is 500 °C. Take the heat transfer coefficient in the oven to be $h = 40$ W/m^2 ·°C. Assume one dimensional heat conduction along the x axis as shown in Fig. Q1.

(a) Based on one-term approximation, find the mid plane temperature of the steel plate as it comes out of the oven, and **[8 marks]**

(b) calculate the time required for the heating process of the plate. **[5 marks]**

The one-term approximation for a plane wall of thickness $2L$, is given as $\theta^* = \theta_o^* \cos(\zeta_1 x^*)$, where θ_o^* represents the temperature at the mid plane and it is of the form $\theta_o^* = C_1 \exp(-\zeta_1^2 Fo)$ where $Fo = \frac{\alpha t}{L^2}$.

The dimensionless variables are defined as $\theta^* = \frac{T - T_\infty}{T_i - T_\infty}$, $x^* = \frac{x}{L}$ where x is the distance measured from the mid plane of the plate.

(c) What is the heat flux to the area normal to x axis of the plate at the time calculated in (b)? **[3 marks]**

(d) Estimate the answer for (b) by using lumped capacitance method. Comment on the answer. **[7 marks]**

(e) If the heat transfer coefficient in the oven is less than 40 W/m^2 ·°C, as stated earlier, what is its effect on the answer in (b)? **[2 marks]**

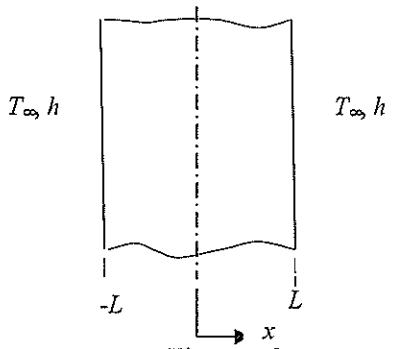

Figure Q1

Note: You may need find Appendix 1 useful for this question.

Continued...

Question 2

(a) The spectral transmissivity of a glass cover used in a solar collector is given as:

$$\tau_\lambda = \begin{cases} 0.7 & 0 < \lambda < 3\ \mu\text{m} \\ 0.2 & \lambda > 3\ \mu\text{m} \end{cases}$$

The solar irradiation on the glass cover is 800 W/m^2 and the absorber plate can be assumed as a blackbody. Assume that the sun emits as a blackbody at 5800 K. Determine the solar irradiation on the absorber plate. **[8 marks]**

Note: You may find Appendix 2 useful for part (a).

(b) An infinitely long duct has a cross-sectional shape of a right-angled triangle with two of its sides equal in width, as indicated in Figure Q2. The temperatures of the two equal sides, each 0.5 m in width, are kept at 400 K and 300 K respectively, while the third side is at temperature 550 K. Sides 1 and 3 may be approximated as blackbodies while side 2 has an emissivity, ε_2 =0.8. Neglect the end effects of the duct.

(i) Determine view factor F_{12}. **[4 marks]**

(ii) Draw the circuit network to represent this problem. **[3 marks]**

(iii) Determine the heat transfer rate per unit length from surface A_3. **[10 marks]**

Take Stefan-Boltzmann constant to be 5.67 x 10 $^{-8}$ W/m^2·K^4

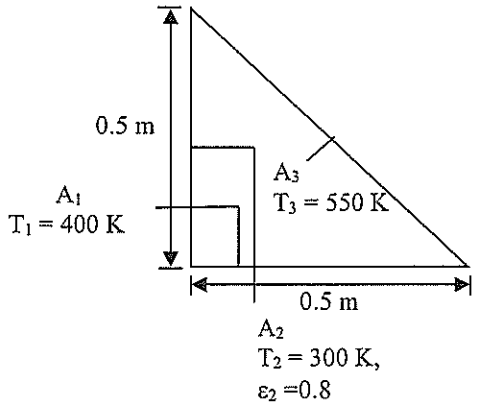

Figure Q2

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Question 3

Boards consisting of heat dissipating components, are attached to the inner surfaces of a rectangular channel, with a cross section of a x b and length L. The boards are cooled by passing air at mass flow rate $\dot{m}$ and inlet temperature T_i through the channel that exits at temperature T_e. Assuming that a constant surface temperature T_s is maintained at the surface of the channel and the air flow is fully developed.

(a) Starting from the steady state energy balance for the channel of width dx, obtain a relationship between the exit temperature of the air, T_e with L, a, b, T_s, T_i, $\dot{m}$, the heat transfer coefficient, h as well as appropriate thermophysical properties. **[7 marks]**

(b) For L =700 mm, a = 350 mm, b = 50 mm, $\dot{m}$ = 0.05 kg/s , T_i= 30 °C and T_e = 40 °C, determine:

(i) the heat transfer rate to the air, q , **[5 marks]**

(ii) the Reynolds number of the air flow, **[7 marks]**

(iii) the heat transfer coefficient for convection in the channel, and **[3 marks]**

(iv) the surface temperature of the channel, T_s . **[3 marks]**

The average Nusselt number correlation for the fully developed turbulent flow in smooth tubes is:

$Nu_D = 0.023 Re_D^{0.8} Pr^n$, n = 0.4 for *heating* and 0.3 for *cooling*.

Note: (i) You may find Appendix 3 useful for this question.

(ii) Hydraulic diameter, $D_H = \frac{4Ac}{P}$.

Continued...

Question 4

(a) (i) Show that the heat transfer rate, q_f from a fin with negligible heat transfer at the tip of the fin is $\sqrt{hPkA_c}\,(T_b - T_\infty)\tanh\left(\sqrt{\frac{hP}{kA_c}}L\right)$. The temperature distribution for such fin is known as $\frac{T-T\infty}{Tb-T\infty} = \frac{\cosh\left(\sqrt{\frac{hP}{kA_c}}(L-x)\right)}{\cosh\left(\sqrt{\frac{hP}{kA_c}}L\right)}$.

All symbols carry the usual meanings. **[4 marks]**

(ii) By letting $L \to \infty$, derive the corresponding heat transfer rate for an infinitely long fin form part (i). **[2 marks]**

Note: You may need these equations in part (b).

(b) An aluminum fin, of length L=9.0 cm, protrudes from a wall. The fin has a uniform circular cross section of diameter 1 cm; its thermal conductivity is k = 200 W/m·K. The base of the fin is maintained at 300 °C, and the ambient temperature is 50 °C. The heat transfer coefficient between the fin and the ambient is h = 10 W/m^2·K. Assume adiabatic tip. An engineer intends to compare the heat transfer rate of an infinitely long fin and a fin with adiabatic tip. What is the ratio of the heat transfer rate of an infinitely long fin to that of a fin with an adiabatic tip in this case? **[4 marks]**

(c) In a one shell and two tube passes heat exchanger, water enters at 10 °C and a rate of 4.2 kg/s to the tube side while hot oil which enters the heat exchanger at 120 °C and a rate of 4 kg/s, is to be cooled in the heat exchanger in the shell side. If the total heat exchange area is given as 8 m^2 and the overall heat transfer coefficient is 1250 W/m^2·°C, determine:

(i) whether water or oil has a minimum thermal capacity, **[2 marks]**
(ii) the NTU for the heat exchange, **[3 marks]**
(iii) the cold water and hot oil outlet temperatures. **[10 marks]**

Consider the specific heats for the water and oil as 4.186 kJ/kg ·°C and 2.2 kJ/kg ·°C respectively.

Note: You may find Appendix 4 useful for part (c).

Continued...

Appendix 1

Coefficients used in the one-term approximation to the series solutions for transient one-dimensional conduction

	Plane Wall		Infinite Cylinder		Sphere	
Bi^a	ζ_1 (rad)	C_1	ζ_1 (rad)	C_1	ζ_1 (rad)	C_1
0.01	0.0998	1.0017	0.1412	1.0025	0.1730	1.0030
0.02	0.1410	1.0033	0.1995	1.0050	0.2445	1.0060
0.03	0.1723	1.0049	0.2440	1.0075	0.2991	1.0090
0.04	0.1987	1.0066	0.2814	1.0099	0.3450	1.0120
0.05	0.2218	1.0082	0.3143	1.0124	0.3854	1.0149
0.06	0.2425	1.0098	0.3438	1.0148	0.4217	1.0179
0.07	0.2615	1.0114	0.3709	1.0173	0.4551	1.0209
0.08	0.2791	1.0130	0.3960	1.0197	0.4860	1.0239
0.09	0.2956	1.0145	0.4195	1.0222	0.5150	1.0268
0.10	0.3111	1.0161	0.4417	1.0246	0.5423	1.0298
0.15	0.3779	1.0237	0.5376	1.0365	0.6609	1.0445
0.20	0.4328	1.0311	0.6170	1.0483	0.7593	1.0592
0.25	0.4801	1.0382	0.6856	1.0598	0.8447	1.0737
0.30	0.5218	1.0450	0.7465	1.0712	0.9208	1.0880
0.4	0.5932	1.0580	0.8516	1.0932	1.0528	1.1164
0.5	0.6533	1.0701	0.9408	1.1143	1.1656	1.1441
0.6	0.7051	1.0814	1.0184	1.1345	1.2644	1.1713
0.7	0.7506	1.0919	1.0873	1.1539	1.3525	1.1978
0.8	0.7910	1.1016	1.1490	1.1724	1.4320	1.2236
0.9	0.8274	1.1107	1.2048	1.1902	1.5044	1.2488
1.0	0.8603	1.1191	1.2558	1.2071	1.5708	1.2732
2.0	1.0769	1.1785	1.5994	1.3384	2.0288	1.4793
3.0	1.1925	1.2102	1.7887	1.4191	2.2889	1.6227
4.0	1.2646	1.2287	1.9081	1.4698	2.4556	1.7202
5.0	1.3138	1.2402	1.9898	1.5029	2.5704	1.7870
6.0	1.3496	1.2479	2.0490	1.5253	2.6537	1.8338
7.0	1.3766	1.2532	2.0937	1.5411	2.7165	1.8673
8.0	1.3978	1.2570	2.1286	1.5526	1.7654	1.8920
9.0	1.4149	1.2598	2.1566	1.5611	2.8044	1.9106
10.0	1.4289	1.2620	2.1795	1.5677	2.8363	1.9249
20.0	1.4961	1.2699	2.2881	1.5919	2.9857	1.9781
30.0	1.5202	1.2717	2.3261	1.5973	3.0372	1.9898
40.0	1.5325	1.2723	2.3455	1.5993	3.0632	1.9942
50.0	1.5400	1.2727	2.3572	1.6002	3.0788	1.9962
100.0	1.5552	1.2731	2.3809	1.6015	3.1102	1.9990
∞	1.5708	1.2733	2.4050	1.6018	3.1415	2.0000

[a] $Bi = hL/k$ for the plane wall and hr_o/k for the infinite cylinder and sphere.

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Appendix 2: Numerical values of the function $F(0 \rightarrow \lambda)$

λT (μm · K)	$F_{(0 \rightarrow \lambda)}$	λT (μm · K)	$F_{(0 \rightarrow \lambda)}$
200	0.000000	9,500	0.903085
400	0.000000	10,000	0.914199
600	0.000000	10,500	0.923710
800	0.000016	11,000	0.931890
1,000	0.000321	11,500	0.939959
1,200	0.002134	12,000	0.945098
1,400	0.007790	13,000	0.955139
1,600	0.019718	14,000	0.962898
1,800	0.039341	15,000	0.969981
2,000	0.066728	16,000	0.973814
2,200	0.100888	18,000	0.980860
2,400	0.140256	20,000	0.985602
2,600	0.183120	25,000	0.992215
2,800	0.227897	30,000	0.995340
3,000	0.273232	40,000	0.997967
3,200	0.318102	50,000	0.998953
3,400	0.361735	75,000	0.999713
3,600	0.403607	100,000	0.999905
3,800	0.443382		
4,000	0.480877		
4,200	0.516014		
4,400	0.548796		
4,600	0.579280		
4,800	0.607559		
5,000	0.633747		
5,200	0.658970		
5,400	0.680360		
5,600	0.701046		
5,800	0.720158		
6,000	0.737818		
6,200	0.754140		
6,400	0.769234		
6,600	0.783199		
6,800	0.796129		
7,000	0.808109		
7,200	0.819217		
7,400	0.829527		
7,600	0.839102		
7,800	0.848005		
8,000	0.856288		
8,500	0.874608		
9,000	0.890029		

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Appendix 3: Properties of air at 1 atm pressure

Properties of air at 1 atm pressure

Temp., T, °C	Density, ρ, kg/m^3	Specific Heat, C_p, J/kg · °C	Thermal Conductivity, k, W/m · °C	Thermal Diffusivity, α, m^2/s	Dynamic Viscosity, μ, kg/m · s	Kinematic Viscosity, ν, m^2/s	Prandtl Number, Pr
−150	2.866	983	0.01171	4.158×10^{-6}	8.636×10^{-6}	3.013×10^{-6}	0.7246
−100	2.038	966	0.01582	8.036×10^{-6}	1.189×10^{-6}	5.837×10^{-6}	0.7263
−50	1.582	999	0.01979	1.252×10^{-5}	1.474×10^{-5}	9.319×10^{-6}	0.7440
−40	1.514	1002	0.02057	1.356×10^{-5}	1.527×10^{-5}	1.008×10^{-5}	0.7436
−30	1.451	1004	0.02134	1.465×10^{-5}	1.579×10^{-5}	1.087×10^{-5}	0.7425
−20	1.394	1005	0.02211	1.578×10^{-5}	1.630×10^{-5}	1.169×10^{-5}	0.7408
−10	1.341	1006	0.02288	1.696×10^{-5}	1.680×10^{-5}	1.252×10^{-5}	0.7387
0	1.292	1006	0.02364	1.818×10^{-5}	1.729×10^{-5}	1.338×10^{-5}	0.7362
5	1.269	1006	0.02401	1.880×10^{-5}	1.754×10^{-5}	1.382×10^{-5}	0.7350
10	1.246	1006	0.02439	1.944×10^{-5}	1.778×10^{-5}	1.426×10^{-5}	0.7336
15	1.225	1007	0.02476	2.009×10^{-5}	1.802×10^{-5}	1.470×10^{-5}	0.7323
20	1.204	1007	0.02514	2.074×10^{-5}	1.825×10^{-5}	1.516×10^{-5}	0.7309
25	1.184	1007	0.02551	2.141×10^{-5}	1.849×10^{-5}	1.562×10^{-5}	0.7296
30	1.164	1007	0.02588	2.208×10^{-5}	1.872×10^{-5}	1.608×10^{-5}	0.7282
35	1.145	1007	0.02625	2.277×10^{-5}	1.895×10^{-5}	1.655×10^{-5}	0.7268
40	1.127	1007	0.02662	2.346×10^{-5}	1.918×10^{-5}	1.702×10^{-5}	0.7255
45	1.109	1007	0.02699	2.416×10^{-5}	1.941×10^{-5}	1.750×10^{-5}	0.7241
50	1.092	1007	0.02735	2.487×10^{-5}	1.963×10^{-5}	1.798×10^{-5}	0.7228
60	1.059	1007	0.02808	2.632×10^{-5}	2.008×10^{-5}	1.896×10^{-5}	0.7202
70	1.028	1007	0.02881	2.780×10^{-5}	2.052×10^{-5}	1.995×10^{-5}	0.7177
80	0.9994	1008	0.02953	2.931×10^{-5}	2.096×10^{-5}	2.097×10^{-5}	0.7154
90	0.9718	1008	0.03024	3.086×10^{-5}	2.139×10^{-5}	2.201×10^{-5}	0.7132
100	0.9458	1009	0.03095	3.243×10^{-5}	2.181×10^{-5}	2.306×10^{-5}	0.7111
120	0.8977	1011	0.03235	3.565×10^{-5}	2.264×10^{-5}	2.522×10^{-5}	0.7073
140	0.8542	1013	0.03374	3.898×10^{-5}	2.345×10^{-5}	2.745×10^{-5}	0.7041
160	0.8148	1016	0.03511	4.241×10^{-5}	2.420×10^{-5}	2.975×10^{-5}	0.7014
180	0.7788	1019	0.03646	4.593×10^{-5}	2.504×10^{-5}	3.212×10^{-5}	0.6992
200	0.7459	1023	0.03779	4.954×10^{-5}	2.577×10^{-5}	3.455×10^{-5}	0.6974
250	0.6746	1033	0.04104	5.890×10^{-5}	2.760×10^{-5}	4.091×10^{-5}	0.6946
300	0.6158	1044	0.04418	6.871×10^{-5}	2.934×10^{-5}	4.765×10^{-5}	0.6935
350	0.5664	1056	0.04721	7.892×10^{-5}	3.101×10^{-5}	5.475×10^{-5}	0.6937
400	0.5243	1069	0.05015	8.951×10^{-5}	3.261×10^{-5}	6.219×10^{-5}	0.6948
450	0.4880	1081	0.05298	1.004×10^{-4}	3.415×10^{-5}	6.997×10^{-5}	0.6965
500	0.4565	1093	0.05572	1.117×10^{-4}	3.563×10^{-5}	7.806×10^{-5}	0.6986
600	0.4042	1115	0.06093	1.352×10^{-4}	3.846×10^{-5}	9.515×10^{-5}	0.7037
700	0.3627	1135	0.06581	1.598×10^{-4}	4.111×10^{-5}	1.133×10^{-4}	0.7092
800	0.3289	1153	0.07037	1.855×10^{-4}	4.362×10^{-5}	1.326×10^{-4}	0.7149
900	0.3008	1169	0.07465	2.122×10^{-4}	4.600×10^{-5}	1.529×10^{-4}	0.7206
1000	0.2772	1184	0.07868	2.398×10^{-4}	4.826×10^{-5}	1.741×10^{-4}	0.7260
1500	0.1990	1234	0.09599	3.908×10^{-4}	5.817×10^{-5}	2.922×10^{-4}	0.7478
2000	0.1553	1264	0.11113	5.664×10^{-4}	6.630×10^{-5}	4.270×10^{-4}	0.7539

Note: For ideal gases, the properties C_p, k, μ, and Pr are independent of pressure. The properties ρ, ν, and α at a pressure P (in atm) other than 1 atm are determined by multiplying the values of ρ at the given temperature by P and by dividing ν and α by P.

Source: Data generated from the EES software developed by S. A. Klein and F. L. Alvarado. Original sources: Keenan, Chao, Keyes, Gas Tables, Wiley, 198; and Thermophysical Properties of Matter, Vol. 3: Thermal Conductivity, Y. S. Touloukian, P. E. Liley, S. C. Saxena, Vol. 11: Viscosity, Y. S. Touloukian, S. C. Saxena, and P. Hestermans, IFI/Plenun, NY, 1970. ISBN 0-305067020-8.

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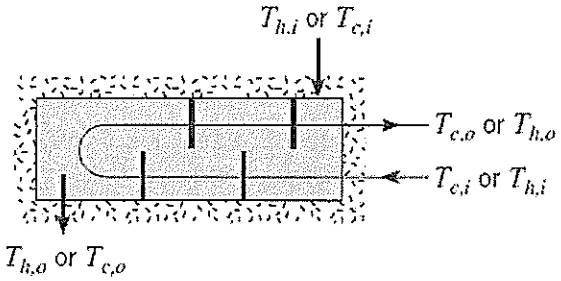
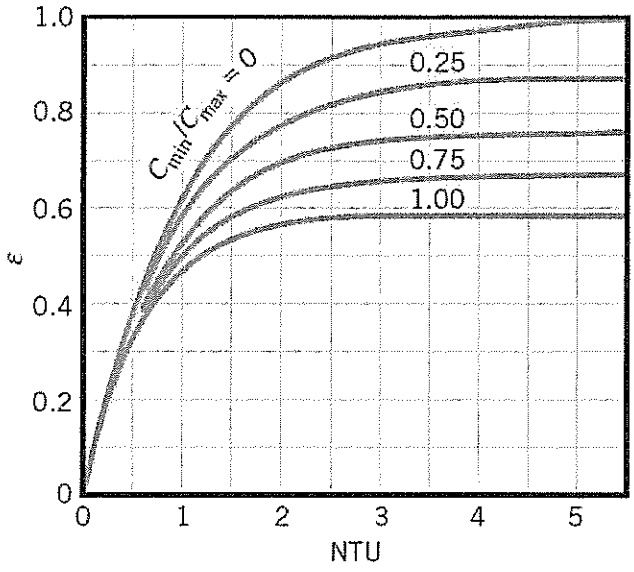
APPENDIX 4: Effectiveness of a shell-and-tube heat exchanger





Effectiveness of a shell-and-tube heat exchanger with one shell and any multiple of two tube passes .

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